FLYWHEEL [CRANKSHAFT] ASSEMBLY FOR INTERNAL COMBUSTION ENGINE

E1

This application is a continuation of application Ser. No. 07/485,659 filed Feb. 27, 1990 now abandoned.

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to a crankshaft assembly including a flywheel, for an internal combustion engine. More specifically, the present invention relates to a crankshaft assembly for an internal combustion engine, which can effectively shift a resonance frequency of a flexural or bending vibration of the crankshaft assembly out of a target frequency band of a forced vibration which results such as during acceleration of a vehicle so as to effectively prevent occurrence of a thick sound or noise in an engine room, while ensuring a quick response for clutch engaging and disengaging operations, and/or which can prevent occurrence of a fore and aft vibration of a vehicle floor at the time of engagement of the clutch.

2. Description of the Background Art

In a known crankshaft assembly for an internal combustion engine, a flywheel is directly connected to a crankshaft to use a mass of the flywheel mainly for reducing a torsional vibration generated in a rotating direction of the crankshaft assembly due to periodic torque fluctuation. However, the mass of the flywheel tends to generate a flexural or bending vibration in an axial direction of the crankshaft which causes a thick sound or noise in an engine room and thus in a vehicle compartment for an automotive vehicle, particularly at the time of the acceleration of the vehicle.

Accordingly, there has been proposed a crankshaft assembly such as disclosed in Second Japanese Patent Publication No. 57-58542, wherein the flywheel is connected to the crankshaft through an elastic or flexible plate. The elastic plate has a rigidity in its rotating direction large enough for effectively transmitting the power between the crankshaft and a transmission through a clutch, while the elastic plate has a rigidity in the axial direction small enough for shifting a resonance frequency of the bending vibration out of a frequency band of a forced vibration which results during the most frequently used engine speed (4,000 rpm) so as to overcome the above-noted problem.

However, the background art as mentioned above has the following problems.

When the rigidity of the elastic plate in the axial direction (hereinafter referred to as "the axial rigidity") is too small, a clutch stroke for engaging and disengaging the clutch is likely to become larger, resulting in a delayed response of the clutch engaging and disengaging operations leading particularly to failure of the clutch disengagement which is likely to cause such as an engine stall. On the other hand, when the axial rigidity of the elastic plate is too large, the deviation of the resonance frequency of the bending vibration from the target frequency band of the forced vibration can not be ensured.

Further, in the background art, when the flywheel is rotated, an axial run-out occurs on an engaging surface of the flywheel with a clutch facing of a clutch disc provided adjacent to the flywheel, due to a processing error and an assembling error of the elastic plate and the flywheel. Accordingly, when the clutch is engaged, a vibration is generated by a combination of the run-out of the engaging surface of the flywheel and the torque fluctuation of the

engine, which is amplified by a vibration generated by the combustion in the engine cylinders and corresponding movements of associated members so as to cause a fore and aft vibration of the vehicle floor. Such vibration is uncomfortable for the driver and passengers in the vehicle compartment.

SUMMARY OF THE INVENTION

Therefore, it is an object of the present invention to provide a crankshaft assembly for an internal combustion engine that can eliminate the above-noted defects inherent in the background art.

It is another object of the present invention to provide a crankshaft assembly for an internal combustion engine that can effectively shift a resonance frequency of a flexural or bending vibration of the crankshaft assembly out of a target frequency band of a forced vibration, particularly out of a target frequency band which results during acceleration of a vehicle so as to effectively prevent occurrence of a thick sound or noise in an engine room, while ensuring a quick response of the clutch engagement and disengagement operations so as to prevent particularly the failure of the clutch disengagement which is likely to cause such as an engine stall.

It is still another object of the present invention to provide a crankshaft assembly for an internal combustion engine that can prevent occurrence of a fore and aft vibration of a vehicle floor at the time of the engagement of the clutch by effectively eliminating an axial run-out of an engaging surface of a flywheel with a clutch facing generated during rotation of the flywheel.

To accomplish the above mentioned and other objects, according to one aspect of the present invention, a crankshaft assembly for an internal combustion engine comprises a crankshaft for transmitting a driving power to a transmission through a clutch, an elastic member fixed to the crankshaft, and a flywheel fixed to the elastic member such that the flywheel is supported in an elastic relationship with the crankshaft.

The flywheel has an engageable surface at a side opposite to the elastic member in an axial direction of the crankshaft, and the engageable surface is engageable with an associated member of the clutch to receive a load therefrom in the axial direction when the engageable surface is engaged with the associated member of the clutch.

The elastic member has a first predetermined rigidity in its rotating direction, the first predetermined rigidity being large enough to effectively transmit the driving power to the transmission through the clutch. On the other hand, the elastic member has a second predetermined rigidity in the axial direction, the second predetermined rigidity being small enough to shift a resonance frequency of a bending vibration out of a target frequency band of a forced vibration, while ensuring to prevent a failure of disengagement between the engageable surface of the flywheel and the associated member of the clutch.

According to another aspect of the present invention, a method for forming a crankshaft assembly for an internal combustion engine comprises steps of fixing a flywheel to an elastic member to form a unit, assembling the unit onto the crankshaft with the elastic member mounted onto the crankshaft so as to support the flywheel in an elastic relationship with the crankshaft, and processing an engageable surface of the flywheel, which is engageable with an associated member of a clutch, based on an assembled condition between the

elastic member and the crankshaft so as to minimize an axial run-out of the engageable surface.

According to still another aspect of the present invention, a crankshaft assembly for an internal combustion engine comprises a crankshaft for transmitting a driving power to a transmission through a clutch, an elastic member fixed to the crankshaft, and a flywheel fixed to the elastic member such that the flywheel is supported in an elastic relationship with the crankshaft.

The flywheel has an engageable surface at a side opposite to the elastic member in an axial direction of the crankshaft, and the engageable surface is engageable with an associated member of the clutch to control transmission of the driving power between the crankshaft and the transmission.

The engageable surface is designed to have an axial run-out which is no more than 0.1 mm for ensuring a smooth engagement with the associated member of the clutch.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention will be understood more fully from the detailed description given hereinbelow and from the accompanying drawings of the preferred embodiment of the invention, which are given by way of example only, and are not intended to be limitative of the present invention.

In the drawings:

FIG. 1 is a longitudinal cross section of a crankshaft assembly for an internal combustion engine according to a first preferred embodiment of the present invention;

FIG. 2 is a graph of vibration level versus frequency showing a shift of a resonance frequency of a flexural or bending vibration by changing a rigidity of an elastic or flexible plate in an axial direction of a crankshaft;

FIG. 3 is a longitudinal cross section of a crankshaft for assembly $\boxed{\text{fop}}$ an internal combustion engine according to a second preferred embodiment of the present invention; and

FIG. 4 is a graph of fore and aft vibration of vehicle floor versus flywheel run-out amount, showing a relationship between an amount of an axial run-out of a flywheel and a fore and aft vibration of a vehicle floor.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Now, a [crankshaft] flywheel assembly for an internal combustion engine according to preferred embodiments of the present invention will be described hereinbelow with reference to FIGS. 1 to 4.

FIG. 1 shows a first preferred embodiment of the present invention. An engine crankshaft 1 is connected to pistons through respective connecting rods in a known manner for receiving the driving power therefrom. An elastic plate 2 of this example is substantially of a disc shape, and is fixed, at its inner portion 2f, to one shaft end of the crankshaft 1 by a plurality of bolts 3. The elastic plate 2 [is formed at its] has an outer peripheral [edge] portion 2b which is formed with an axially extending [section] flange 2a to which a ring gear R is fixed. The ring gear R engages with pinion gears of an engine starter motor for transmitting the driving power from the engine starter motor to the crankshaft 1 when starting the engine.

An annular reinforcing member 4 is disposed between the elastic plate 2 and heads of the bolts 3. The reinforcing member 4 is formed at its outer peripheral edge portion with a received portion 4a which is in this example cylindrical [section 4a] and [extending] extends in an axial direction of the crankshaft 1. [and with] The reinforcing member 4 of this example further has a radially outwardly extending [section] flange 4b in the form of an outward flange, as shown in Fig. 1. The inner portion 2f of the elastic plate 2 is clamped between the reinforcing member 4 and the shaft end of the crankshaft 1.

Ins H2

A flywheel body 5 of an annular shape is fixed to the elastic plate 2 at their respective outer peripheral [edge] portions 5a and 2b through a plurality of bolts 6 and corresponding reinforcing ring members 7 disposed between the elastic plate 2 and heads of the bolts 6. The annular flywheel body 5 has an inner portion 5h [a stepped inner peripheral edge surface] defining a central mounting [opening] hole 5b for receiving the cylindrical received portion 4a of the reinforcing member 4 therein. The [stepped] inner peripheral [edge] surface of the flywheel body 5 is stepped and has a first surface section 5c extending axially, a second surface section 5d extending radially outward from the first surface section 5c and a third surface section 5e extending axially from the second surface section 5d. The [axial section] axially extending, cylindrical received portion 4a of the reinforcing member 4 is in a slidable contact with the first surface section 5c of the flywheel body 5, and the radial [section] outward flange 4b of the reinforcing member 4 is spaced from the second surface section 5d of the flywheel body 5 by a predetermined [distance] clearance 10 for allowing an axial movement of the flywheel body 5 along with the elastic plate 2. A radially extending [inner] first side surface 5f of the flywheel body 5 facing the elastic plate 2 is spaced apart from the elastic plate 2 by a predetermined [distance] clearance 11 for ensuring an elasticity of the elastic plate 2.

The flywheel body 5 further includes a radially extending side surface 5g at a side axially opposite to the side [radial] surface 5f or the elastic plate 2. The [radial] radially extending side surface 5g is an engaging surface which is engageable with a clutch facing 8 of a clutch disc 9 of a clutch in a known manner so as to control the transmission of the power between the crankshaft 1 and a transmission.

A rigidity of the elastic plate 2 in its rotating direction (hereinafter referred to as "the circumferential rigidity") is set large enough for effectively transmitting the power between the crankshaft 1 and the transmission through the clutch, while a rigidity of the elastic plate 2 in the axial direction (hereinafter referred to as "the axial rigidity") is set small enough for shifting a resonance frequency of the flexural or bending vibration out of a frequency band of a forced vibration which results during the acceleration of the engine.

As described in the background art, when the axial rigidity of the elastic plate is too small, a clutch stroke for engaging and disengaging the clutch becomes larger, i.e. a clutch stroke loss gets larger, resulting in delayed response of the clutch engaging and disengaging operations leading particularly to the failure of the clutch disengagement which is likely to cause such as an engine stall. On the other hand, when the axial rigidity of the elastic plate is too large, the deviation of the resonance frequency of the bending vibration from the target frequency band of the forced vibration can not be attained.

To overcome the above-noted problem, the axial rigidity of the elastic plate 2 in this embodiment is set to 600 kg/mm to 2200 kg/mm, wherein an axial displacement of the radial surface 5*g* of the flywheel 5 is no more than 1 mm when an axial load or force 600 kg to 2200 kg is applied to the radial surface 5*g*. By selecting a value of the axial rigidity of the elastic plate 2 within the foregoing range, not only is the failure of the clutch disengagement effectively prevented, but also the deviation of the resonance frequency of the bending vibration from the frequency band of the forced vibration, during the acceleration of the engine in this embodiment, is effectively attained so as to prevent generation of the thick sound or noise in the engine room.

Specifically, it is confirmed that the failure of the clutch disengagement, i.e. the failure of the disengagement between the radial surface 5*g* of the flywheel and the clutch facing 8 of the clutch disc 9, happens when an axial displacement of the radial surface 5*g* at the time of engage-

ment with the clutch facing 8 exceeds 5% of a normal clutch stroke (normally at 7 mm to 8 mm) for [fur] engaging and disengaging the clutch. The normal clutch stroke is a distance between the radial surface 5g of the flywheel body 5 and the clutch facing 8 in a disengagement or released condition of the clutch. Accordingly, considering that an axial load applied to the flywheel body 5 through the clutch facing 8 is normally at 150 kg to 200 kg, the lower limit value 600 kg/mm of the axial rigidity of the elastic plate is selected, wherein the axial displacement of the radial surface 5g is within 5% of the normal clutch stroke when applied with the axial load 150 kg to 200 kg, as shown in TABLE 1.

TABLE 1

AXIAL LOAD	AXIAL RIGIDITY	AXIAL DISPLACEMENT
150 kg	500 kg/mm	0.30 mm (3.8 to 4.3%)
200 kg	500 kg/mm	0.40 mm (5.0 to 5.7%)
150 kg	600 kg/mm	0.25 mm (3.1 to 3.6%)
200 kg	600 kg/mm	0.33 mm (4.1 to 4.7%)
150 kg	700 kg/mm	0.21 mm (2.6 to 3.0%)
200 kg	700 kg/mm	0.29 mm (3.6 to 4.1%)

(wherein, percentage denotes a rate of the axial displacement relative to the normal clutch stroke which is 7 to 8 mm)

As seen from TABLE 1, the lower limit value 600 kg/mm of the axial rigidity of the elastic plate 2 ensures the axial displacement of the radial surface 5g of the flywheel body 5 within 5% of the normal clutch stroke, i.e. the axial displacement of the radial surface 5g is between 0.25 to 0.33 mm or between 3.1 to 4.7% relative to the normal clutch stroke when applied with the normal axial load at 150 to 200 kg through the clutch facing 8, so that the failure of the clutch disengagement is effectively prevented. Naturally, the larger the axial rigidity of the elastic plate gets, the smaller the axial displacement of the flywheel gets.

Now, the axial rigidity of the elastic plate 2 will be considered in view of shifting of the resonance frequency of the bending vibration out of a frequency band of a forced vibration which results during the acceleration of the engine where the sound or noise generated by the bending vibration is the most significant. It is confirmed that the sound or noise generated by the bending vibration is effectively reduced when the resonance frequency is shifted out of the frequency band of the forced vibration during the acceleration of the engine.

FIG. 2 is a graph of bending vibration level versus frequency showing a result of experiments using various elastic plates having different axial rigidities. The frequency band of the forced vibration during the acceleration of the engine is 200 Hz to 500 Hz. In FIG. 2, a line Ao shows a relationship between the frequency and the bending vibration level without using the elastic plate, i.e. the flywheel is directly connected to the crankshaft. As can be seen, a resonance frequency of the line Ao is within 200 Hz to 500 Hz, which causes the sound or noise problem. A line A1 is derived by the elastic plate having the axial rigidity of 2200 kg/mm, a line A2 is derived by the elastic plate having the axial rigidity of 1700 kg/mm, line A3 is derived by the elastic plate having the axial rigidity of 1200 kg/mm, and a line A4 is derived by the elastic plate having the axial

rigidity of 1000 kg/mm. As can be seen, the resonance frequency of each of the lines A1 to A4 is shifted out of the frequency band 200 Hz to 500 Hz, and further, the vibration level of each of the lines A1 to A4 is considerably lower than the line Ao within the frequency band 200 Hz to 500 Hz. Though the line A1 has a vibration level higher than the line Ao around 200 Hz, this happens in a very small range of frequency. Accordingly, the value 2200 kg/mm is selected as an upper limit value of the axial rigidity of the elastic plate, and the value 1700 kg/mm is selected as a more preferable upper limit value of the axial rigidity.

In light of the above, the axial rigidity of the elastic plate 2 in this embodiment is selected at 600 kg/mm to 2200 kg/mm, and preferably at 600 kg/mm to 1700 kg/mm.

As understood from the above description, this first embodiment, when the crankshaft 1 is rotated, the flywheel body 5 is ensured to rotate with the crankshaft 1 by means of the large circumferential rigidity of the elastic plate 2. When the clutch is engaged and the engine is accelerated, the driving power is transmitted to the transmission with a very low bending vibration level by means of the axial rigidity of the elastic plate being no more than 2200 kg/mm, so that the vehicle compartment can be kept quiet. On the other hand, when the clutch is disengaged, since the axial displacement of the flywheel is no more than 5% of the normal clutch stroke by means of the axial rigidity of the elastic plate being no less than 600 kg/mm, the failure of the disengagement of the clutch is effectively prevented.

flywheel

FIG. 3 shows a [crankshaft] assembly for an internal combustion engine according to a second embodiment of the present invention. In FIG. 3, the same or like parts or members are denoted by the same reference numerals. In the following description, explanations of those same or like members will be omitted to avoid redundant description. Further, though the clutch assembly is not shown in FIG. 3, the same clutch assembly including the clutch disc 9 and the clutch facing 8 is provided in the same manner as in FIG. 1.

In FIG. 3, the crankshaft 1 includes a stepped end surface having a first section la extending radially inward from its outer peripheral edge, a second section *1b* extending axially from the inward end of the first section *1a* toward the clutch disc 9, and a third circular section *1c* extending radially from the second section *1b*. The elastic plate 2 is of an annular shape having a mounting opening at its center for receiving the second section *1b* therethrough. The elastic plate 2 is fixed to the crankshaft 1 with its axially extending inward end *2c* facing the second section of the crankshaft 1 and with its radially extending inward end portion *2d* facing the first section of the crankshaft. The other structure is substantially the same as in FIG. 1.

engaging

As mentioned in the background art, when the flywheel body 5 is rotated through the crankshaft 1, an axial run-out is generated on the radial surface *5g* due to the processing error and the assembling error of the elastic plate 2 and the flywheel body 5 to cause the vibration when the clutch is engaged. The vibration further causes the fore and aft vibration of the vehicle floor.

engaging

In order to overcome the above-noted problem, in this embodiment, the radial surface *5g* is processed in a manner to make an amount of the axial run-out no more than 0.1 mm. Specifically, the processing of the radial surface *5g* is performed in the following manner. engaging

The flywheel body 5 is first fixed to the elastic plate 2 by the bolts 6. Then, this unit is assembled to the crankshaft 1 with the axially extending inward end *2c* of the elastic plate 2 facing the second section *1b* of the crankshaft 1 and with

the radially extending inward end portion **2d** facing the first section **1a** of the crankshaft. Then, the radial engaging surface **5g** is processed based on the assembled condition between the axially extending inward end **2c** and the second section **1b** and/or between the radially extending inward end portion **2d** and the first section **1a** to make the axial run-out of the radial surface **5g** no more than 0.1 mm.

By using the above-noted manner, the radial engaging surface **5g** is easily and precisely processed to make the amount of the axial run-out no more than 0.1 mm.

FIG. **4** is a graph of axial run-out amount of flywheel (radial engaging surface **5g**) versus fore and aft vibration of vehicle floor showing a result of experiments. It is confirmed that the fore and aft vibration of the vehicle floor which does not give a uncomfortable feeling to a human body is normally no more than 0.1 G (gravitational acceleration). As can be seen from FIG. **4**, a fore and aft vibration of the vehicle floor is substantially in direct proportion to an amount of the axial run-out of the radial engaging surface **5g**, and the fore and aft vibration becomes no more than 0.1 G when the axial run-out becomes no more than 0.1 mm. Accordingly, by making the amount of the axial run-out no more than 0.1 mm as in this embodiment, the fore and aft vibration can be made no more than 0.1 G.

As understood from the above description, in this second embodiment, when the crankshaft **1** is rotated, the flywheel body **5** is ensured to rotate with the crankshaft **1** by means of the large circumferential rigidity of the elastic plate **2**. Since the amount of the axial run-out of the radial engaging surface **5g** is no more than 0.1 mm, the engagement between the radial engaging surface **5g** and the clutch facing **8** is performed quite smoothly, so that the fore and aft vibration does not exceed 0.1 G. Accordingly, the driving power is transmitted from the engine to the transmission without giving the uncomfortable feeling to the human body.

It is to be appreciated that in this second embodiment, the axial rigidity of the elastic plate **2** is not necessarily selected at 600 kg/mm to 2200 kg/mm.

It is to be understood that the invention is not to be limited to the embodiments described above, and that various changes and modifications may be made without departing from the spirit and scope of the invention as defined in the appended claims.

What is claimed is: